

*Application
for
United States Patent*

To all whom it may concern:

*Be it known that Chuan Weng has invented certain new and useful
improvements in*

***NON-CFC REFRIGERANT MIXTURE FOR AN ULTRA-LOW
TEMPERATURE REFRIGERATION SYSTEM***

of which the following is a full, clear and exact description:

**NON-CFC REFRIGERANT MIXTURE FOR AN ULTRA-LOW
TEMPERATURE REFRIGERATION SYSTEM**

FIELD OF INVENTION

[0001] The present invention relates generally to an apparatus for low temperature refrigeration systems. More particularly, the present invention relates to a substantially non-chlorofluorocarbon (non-CFC) design of a refrigerant mixture for an ultra-low temperature refrigeration system.

BACKGROUND OF THE INVENTION

[0002] In refrigeration systems, a refrigerant gas is compressed in a compressor unit. Heat generated by the compression is then removed generally by passing the compressed gas through a water or air cooled condenser coil. The cooled, condensed gas is then allowed to rapidly expand into an evaporating coil where the gas becomes much colder, thus cooling the coil and the inside of the refrigeration system box around which the coil is placed.

[0003] Ultra-low and cryogenic temperatures ranging from -95° C to -150° C. have been achieved in refrigeration systems using a single circuit vapor compressor. These systems typically use a single compressor to pump a

mixture of four or five chlorofluorocarbon (CFC) containing refrigerants to reach an evaporative temperature of as low as -160° C.

[0004] Environmental concern over the depletion of the ozonosphere has increased pressure on refrigerator manufacturers to substantially reduce the level of CFC-containing refrigerants used within their systems. Although non-CFC refrigerant mixtures have been developed, it has been discovered that most of these refrigerant mixtures cannot simply be substituted for CFC-containing refrigerants in currently available refrigeration systems due to the different thermodynamic properties of the refrigerants.

[0005] The present inventor has discovered that using substantially non-CFC refrigerants in conventional ultra-low and cryogenic temperature systems cause an imbalanced flow of the refrigerants in the refrigeration circuit, which reduces the cooling capability of the refrigerants to the compressor. Such low levels of compressor cooling can cause a system to fail due to compressor overheating.

[0006] Unlike the CFC-containing refrigeration systems which do not cause overheating of the compressor, the present inventor has discovered that the substantially non-CFC refrigeration systems must provide additional liquid return to the compressor in order to avoid overheating thereof and eventual failure of the system.

[0007] The present inventor has been able to overcome the overheating of the compressor when using substantially non-CFC refrigerants in a single

compressor autocascade system. This is accomplished by providing a specially-designed capillary tube or expansion means disposed downstream of the first liquid/gas separator such that liquid refrigerants are returned directly to the auxiliary condenser and then to the compressor. This feature enables larger than normal quantities of refrigerants of higher boiling points to be rapidly returned to the compressor, which results in excellent operating conditions of the compressor and avoids overheating thereof.

[0008] As such, the overall performance of the non-CFC autocascade system is comparable to its counterpart of the CFC autocascade system. This is evidenced by the fact that both systems have similar pull down rates and compressor operating conditions at standard 90° F ambient.

[0009] The present invention also provides many additional advantages which shall become apparent as described below.

SUMMARY OF THE INVENTION

[0010] The present invention overcomes the need for using CFC refrigerant mixtures in a refrigeration system by utilizing refrigerants R14, R134a, R508a or R508b, R142b, and R740 in a component mixture. To achieve desired properties, these refrigerants may be used in a "cocktail" mixture.

[0011] It is therefore a feature of the present invention to provide a substantially non-CFC ultra-low temperature refrigerant mixture that can

safely be applied in the field as needed without the risks associated with CFC or HCFC ultra-low temperature refrigerants.

[0012] It is another feature of the present invention to provide a refrigeration heat exchanger section which is capable of circulating a substantially non-CFC refrigerant mixture which comprises: a compressor means, an auxiliary condenser, a first condenser, a second condenser, a third condenser, a subcooler means and a liquid/gas separator, wherein the improvement is characterized by: a means for distributing a subcooled refrigerant liquid mixture from the liquid/gas separator to a first expansion means and a second expansion means for forming first and second expanded streams, respectively; and a first conduit means for returning the first expanded stream to the auxiliary condenser and the compressor; and a second conduit means for delivering the second expanded stream to the first condenser.

[0013] More specifically, the refrigeration heat exchanger section preferably comprises: a compressor means; an auxiliary condenser connected to receive and cool the refrigerant mixture discharged from the compressor means; a first liquid/gas separator connected to received the cooled refrigerant mixture discharged from the auxiliary condenser, wherein a subcooled refrigerant liquid mixture is taken as bottoms and a gaseous refrigerant liquid mixture is taken overhead; a means for distributing the subcooled refrigerant liquid mixture to a first expansion means and a second expansion means to

form a first expanded stream and a second expanded stream, respectively; a first conduit means for returning the first expanded stream to the auxiliary condenser and the compressor.

[0014] The high pressure flow of the heat exchanger circuit further comprises: a first condenser connected to receive the gaseous refrigerant mixture from the liquid/gas separator; a second liquid/gas separator connected to receive the gaseous refrigerant mixture from the first condenser, wherein a subcooled liquid refrigerant mixture is taken as bottoms and a gaseous refrigerant mixture is taken overhead; a second condenser connected to receive the gaseous refrigerant mixture which is taken overhead from the second liquid/gas separator; a third condenser connected to receive at least a portion of the gaseous refrigerant mixture taken from the second condenser; and a subcooler means connected to receive the gaseous refrigerant mixture from the third condenser.

[0015] The low pressure flow of the heat exchanger circuit further comprises: a distributor means connected to receive the refrigerant mixture from the subcooler means, the distributor means is capable of separating the refrigerant mixture into a first stream and a second stream; a third expansion means connected to receive the first stream, thereby forming a third expanded stream; a third conduit means for delivering the third expanded stream to the subcooler means; a fourth expansion means connected to received the second stream, thereby forming a fourth expanded stream; a fourth conduit means for

delivering the fourth expanded stream to a storage tank; a fifth conduit means for delivering the fourth expanded stream from the storage tank to the third condenser; a sixth conduit means disposed between the third condenser and the second condenser such that the fourth expanded stream from the third condenser is delivered to the second conduit means; a sixth expansion means connected to receive the subcooled liquid refrigerant mixture from the second liquid/gas separator, thereby forming a fifth expanded stream; a seventh conduit means for delivering the fifth expanded stream to the second condenser; an eighth conduit means for delivering the fifth expanded stream from the second condenser to the first condenser; a second conduit means for delivering the second expanded stream to the first condenser; a ninth conduit means for delivering the second expanded stream and the fifth expanded stream from the first condenser to the auxiliary condenser; and a tenth conduit means for delivering the first, second and fifth expanded streams from the auxiliary condenser to the compressor.

[0016] There has been outlined, rather broadly, the more important features of the invention in order that the detailed description thereof that follows may be better understood, and in order that the present contribution to the art may be better appreciated. There are, of course, additional features of the invention that will be described below and which will form the subject matter of the claims appended hereto.

[0017] In this respect, before explaining at least one embodiment of the invention in detail, it is to be understood that the invention is not limited in its application to the details of construction and to the arrangements of the components set forth in the following description or illustrated in the drawings. The invention is capable of other embodiments and of being practiced and carried out in various ways. Also, it is to be understood that the phraseology and terminology employed herein, as well as the abstract, are for the purposes of description and should not be regarded as limiting.

[0018] As such, those skilled in the art will appreciate that the conception upon which this disclosure is based may readily be utilized as a basis for the designing of other structures, methods and systems for carrying out the several purposes of the present invention. It is important, therefore, that the claims be regarded as including such equivalent construction insofar as they do not depart from the spirit and scope of the present invention.

BRIEF DESCRIPTION OF THE DRAWINGS

[0019] FIG. 1 is a block diagram of the single compressor refrigeration system according to the present invention.

[0020] FIG. 2 is a non-CFC autocascade heat exchanger section according to the present invention.

[0021] FIG. 3 is a conventional CFC-based autocascade heat exchanger section.

[0022] FIG. 4 is a graph depicting the Saturation Pressure characteristics for R14 and R740 in an autocascade system at -156° C of the present invention.

[0023] FIG. 5 is a graph depicting the refrigerant mixture characteristics in Temperature vs. Time in an autocascade system according to the present invention.

[0024] FIG. 6 is a graph depicting the stability characteristics of the refrigerant mixture according to the present invention.

[0025] FIG. 7 is a graph depicting the refrigerant mixture pull down rate characteristics in an autocascade system according to the present invention.

[0026] FIG. 8 is a graph depicting the pull down pressures of the refrigerant mixture according to the present invention.

DETAILED DESCRIPTION OF THE INVENTION

[0027] Referring now to the figures, in FIG. 1 there is shown a Single compressor ultra-low and cryogenic temperature refrigeration systems, as shown in FIG. 1, pump refrigerants through a condenser, heat exchanger section and evaporator coils in a closed circuit loop to provide temperatures as low as -150° C. The heat exchanger section and evaporator coils referred to in FIG. 1 are specifically described in FIG. 2. The conventional refrigeration compressor and condenser referred to in FIG. 1 are not shown in FIG. 2. The

air-cooled or water-cooled condenser cools the compressor and removes BTU's from the refrigerant by partially changing the refrigerant mixture from vapor to liquid, whereas the liquid/gas separator separates liquid refrigerant from vapor and returns lubricating oil to the compressor. The heat exchangers use the thermophysical properties of the refrigerants to effect the cooling process. The evaporator coils permit the flow of refrigerant at ultra-low temperatures to absorb heat from the freezer interior, delivering this heat to the condenser for removal.

[0028] For example, a substantially non-CFC refrigerant mixture used with this system is the combination of five refrigerants R134a (CF_3CFH_2) at about 20.8% by volume; R508a or R508b (R23 + R116b) at about 20.4% by volume; R14 (CF_4) at about 18.2% by volume; R142b (CH_3CCl_3) at about 22.8% by volume; and R740 (argon, Ar) at about 17.8% by volume. The -95° C systems use a similar heat exchanger configuration.

[0029] FIG. 2 is a schematic diagram for a -150° C non-CFC autocascade heat exchanger section, wherein a mixture of substantially non-CFC refrigerant is pumped from liquid line 1 taken from the condenser shown in FIG. 1 through heat exchanger 3 to produce a mixture of gases and liquids at 225 psi and room temperature. The heat exchanger 3 combines the functions of a desuperheater and an auxiliary heat exchanger. This liquid/gas mixture produced by heat exchanger 3 is then pumped through auxiliary condenser 7 via conduit 5 and exits therefrom via conduit 9. After flowing

through auxiliary condenser 7 the liquid/gas mixture reaches a temperature of approximately -10° F.

[0030] For example, at -10° F and a pressure of about 220 psi, refrigerants R142b, R134a and R508 become subcooled liquids, and sink to the bottoms of a vertically-mounted liquid/gas separator 11. The subcooled liquid mixture is then distributed and expanded by two capillary tube 13 and 15. The expanded liquid flows from capillary tube 13 and 15 to conduits 17 and 21, respectively, to join the return flow of low pressure refrigerant fluids.

[0031] Meanwhile, R14 and R740, along with traces of the other refrigerants of higher boiling points, continue to flow through the tube side of first condenser 23 via conduit 25. The temperature of the R14 and R740 after passing through first condenser 23 is approximately -67° F. The traces of R508 are subcooled to a liquid phase after passing through first condenser 23 such that it passes from conduits 35 and 37 into liquid/gas separator 39. Liquid R508 and some gases are expanded by capillary tube 41 and pumped via conduits 43 and 45 to the tube side of second condenser 47. After passing through second condenser 47, the liquid R508 is mixed in conduit 27 with the expanded mixture from conduit 21 and returned to the shell side of first condenser 23.

[0032] The R14 and argon gas exiting first condenser 23 via conduit 35 are pumped via conduit 49 to the shell side of second condenser 47, exiting therefrom via conduit 51 at a typical temperature of -130° F. This temperature

and the high side pressure of 215 psig allow a portion of the R14 to be subcooled and sent via conduit 53 to capillary tube 55 where it is expanded and pumped via conduit 57 to cool the tube side of third condenser 59. However, a majority of the R14 and R740 are passed through the shell side of third condenser 59 to conduit 61 and into the tube side of subcooler 63. Most of the R14 and R740 exit subcooler 63 via conduit 65 at a temperature of -220° F. These gases are distributed via conduits 67 and 68 to capillary tube 69 and 70, respectively, where they are expanded to achieve a final temperature of -260° F. The expanded R14 and R740 from capillary tube 70 enter the shell side of subcooler 63 via conduit 72 to cool the gases passing through the tube side of subcooler 63. These gases then exit subcooler 63 via conduit 74 and are joined in conduit 57 with the expanded gases contained in reservoir or storage tank 76 (*i.e.*, this constitutes the evaporator coils of FIG. 1) and expanded gases from capillary tube 55 before passing through the tube side of third condenser 59.

[0033] A portion of the R14 and R740 which exit second condenser 47 via conduit 51 are diverted via conduit 80 to an expansion tank section (not shown) as needed to prevent overpressure of the system during pull down and heavy loading situations.

[0034] Contemporaneously, the expanded liquid from capillary tube 15 is plumped via conduit 21 to conduit 27 wherein it flows to the shell side of first condenser 23. The shell side liquid of first condenser 23 is then merged

with the expanded liquid from conduit 17 in conduit 29 and sent to the shell side of auxiliary condenser 7. The expanded liquid from conduit 29 exits auxiliary condenser 7 via conduit 31 and passes along the shell side of heat exchanger 3 where it is sent via suction line 33 to a single compressor (*i.e.*, shown in FIG. 1). The compressor referred to in FIG. 1 compresses the expanded liquid and delivers the compressed liquid the condenser of FIG. 1 so as to complete the closed loop circuit of FIG. 1. The use of capillary tube 13 allows liquid phase refrigerants R142b and R134a to continue the journey of evaporation within auxiliary condenser 7 and heat exchanger 3, giving an appropriate return condition to prevent the compressor (not shown) from overheating. Simultaneously, capillary tube 15 will dispatch enough liquid for the cooling of first condenser 23. The use of an additional capillary tube 13 to return refrigerants R142b and R134a to the compressor accommodates the different thermodynamic properties of the non-CFC refrigerants. Otherwise, sufficient liquid refrigerants would not be returned to the compressor to avoid overheating, thereby causing failure of the refrigeration system.

[0035] FIG. 3 depicts a conventional CFC-autocascade heat exchanger section which is similar to the non-CFC systems shown in FIG. 2, except that the subcooled liquid from liquid/gas separator 11 is only distributed and expanded via one capillary tube to the shell side of the first condenser for cooling of the first condenser, second condenser and the compressor. As such, the conventional CFC system of FIG. 3 would cause the compressor to

overheat, if used with the non-CFC refrigerants, and eventually result in a system failure.

[0036] Conversely, if a CFC refrigerant is added to the non-CFC autocascade refrigeration systems according to the present invention, then the thermodynamic operation of the system would be completely disrupted by returning too much liquid to the auxiliary condenser and thus causing the compressor to be flooded and eventual failure of the compressor.

[0037] The refrigerant mixture should consist of R142b (22 oz.), R134a (20 oz.), R508b or R508a (18.2 to 19.7 oz.), R14 (16.7 to 17.5 oz.) and R740 (14.6 to 17.1 oz.) to achieve a freezer of -140° C to -154° C.

[0038] FIG. 4 clearly shows that the impact of the composition R740 in the mixture of R740 and R14. In order to achieve a center air of -153° C, the evaporator temperature needs to operate at -156° C. The ideal composition is determined by adding the pressure drop value to a suction pressure on a Pressure-Composition chart, as shown in FIG. 4, which is established to study the relationship of molar fraction and saturation pressure based on the temperature of -156° C for the mixture of R14 and R740. Since a normal operating pressure drop from the evaporator tank to the compressor suction is about 40 psi, and a normal operating suction pressure is about 65 psia, the evaporator operating pressure is then about 105 psia. FIG. 4 shows the theoretical molar fraction of R740 in a R14/R740 mixture should be approximately 0.7.

[0039] FIG. 5 shows how the evaporator temperature changes with ambient. FIG. 5 also shows that the evaporator inlet temperature and operating pressure are oscillating in a small scale. This indicates that a high rate of local heat flux causes a rapid vaporization of R740 liquid. As the R740 vapor bubble forms and collapses, the pressure and temperature change accordingly. Approximately 30 grams of R14 is also added to the system to increase the evaporator temperature by 3° C. The temperature rise makes the evaporator less vulnerable to heat flux. Thus, the flow and temperature are stabilized.

[0040] FIG. 6 shows evaporator temperature and the center air temperature relationships of the refrigeration system. Clearly, the evaporator inlet temperature is about -157° C as desired.

[0041] FIGS. 7 and 8 show the compressor temperature, cabinet temperature and pressures with respect to the refrigerant mixture of the present invention.

[0042] The present invention being capable of achieving -154° C at the bottom out condition at 27° C should demonstrate its capability to achieve a colder cabinet temperature with a larger condenser and longer capillary tube 69 and refrigerant R508b as a charge.

[0043] It should be noted that the lower temperatures at suction, as exhibited in the non-CFC system, are highly desirable since these lower temperatures assist in the cooling of the compressor.

[0044] The above description and drawings are only illustrative of preferred embodiments which achieve the objects, features, and advantages of the present invention, and it is not intended that the present invention be limited thereto. Any modification of the present invention which comes within the spirit and scope of the following claims is considered to be part of the present invention.